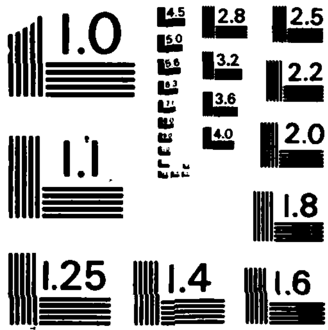
AD-A171 058 INVESTIGATION OF BLADE VIBRATION T55-L-11C COMPRESSOR STAGES 1 AND 2(U) AERONAUTICAL RESEARCH LABS MELBOURNE (AUSTRALIA) N S SWANSSON SEP 85 ARL-AERO-PROP-R-168 1/1

UNCLASSIFIED F/G 21/5 NL



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

(12)

ARL-AERO-PROP-R-168 AR-004-046

AD-A171 058

DEPARTMENT OF DEFENCE

DEFENCE SCIENCE AND TECHNOLOGY ORGANISATION

AERONAUTICAL RESEARCH LABORATORIES

MELBOURNE, VICTORIA

AERO PROPULSION REPORT 168

INVESTIGATION OF BLADE VIBRATION

T55-L-11C COMPRESSOR STAGES 1 AND 2

by

N. S. SWANSSON

Approved for Pubic Release

SEPTEMBER 1985

AR-004-046

DEPARTMENT OF DEFENCE
DEFENCE SCIENCE AND TECHNOLOGY ORGANISATION
AERONAUTICAL RESEARCH LABORATORIES

AERO PROPULSION REPORT 168

INVESTIGATION OF BLADE VIBRATION

T55-L-11C COMPRESSOR STAGES 1 AND 2

by

N. S. SWANSSON

SUMMARY

A vibration survey of the first and second stage compressor blades on the T55-L-11C engine was conducted following the manufacturer's warning that the proposed fitting of locally designed intake screens might lead to blade vibration problems. Vibration tests and a simple analysis indicated a possible resonance region. Special statistical methods for small samples were used to fully define tolerance limits for this region and establish confidence levels in the limits.

Operators were initially advised to restrict operation in the potential resonance region. Subsequent intake flow tests demonstrated that the screens generated no abnormal excitation, so the restrictions could be discontinued.

POSTAL ADDRESS: Director, Aeronautical Research Laboratories,
Box 4331, P.O., Melbourne, Victoria, 3001, Australia

CONTENTS

1. INTRODUCTION

The first and second stage compressor blades on the Lycoming T55-L-11C engine installed in the Boeing Vertol CH-47C Chinook helicopter are liable to foreign object damage (FOD) due to ingested debris. During certain types of operations such as over gravel surfaces, the level of FOD experienced by the RAAF was such as to significantly reduce the operational availability of the helicopter.

To alleviate the FOD and other problems, the engine manufacturers proposed a modification, replacing the original first and second stage compressor blades with a more robust design having a wider chord. While this modification offered a long term solution, pending its implementation the RAAF proposed to cover the standard engine intake screens (which are of relatively coarse mesh) with a fine mesh fibreglass overlay.

Since the proposal to fit overlay screens originated locally, the RAAF asked ARL to investigate the effects of fitting screens on various aspects of engine performance and operation, particularly since the engine manufacturer had warned of a possible problem with "second order" blade vibration [1].

Simple investigations verified that fitting screens would have only minor effects upon engine performance. But as no detailed blade vibration data were available from the engine manufacturer, investigations were initiated both to measure the effect of the screens on vibration excitation due to flow distortion at the compressor intake [2], and to measure blade vibration and response characteristics.

2. TESTS AND RESULTS — ORIGINAL DESIGN BLADES

It was originally expected that the vibration investigation would comprise simple static measurements of natural frequencies, which would then be adjusted for effects of rotation. But the scatter found in frequency values indicated that a greater depth of statistical analysis was called for, in order to have adequate confidence in predictions which would be based on scatter limits.

2.1 Testing

Natural frequencies of blades were determined by measuring resonant frequencies, as the blades were excited through a range of frequencies using an electro magnetic vibrator. Blades were held in a fixture designed to simulate the constraint in a rotating engine, where centrifugal force firmly wedges the blade base into its mating dovetail slot. Peak vibration amplitudes were determined using a non-contacting capacitive displacement pick-up, and mode shapes were observed using stroboscopic illumination, with the stroboscope triggered by a slow motion generating device. Results for a sample group of blades, including mean frequencies and standard deviation estimates, are given in Table 1.

2.2 Effect of Rotation

As the compressor rotates, centrifugal forces tend to straighten the blade, effectively increasing its stiffness and hence its natural frequency. The quantitative effect of centrifugal stiffening can be determined using a Southwell type formula, which is exact for a linear vibrating system

TABLE 1

Natural Frequencies
T55 Compressor — Original Blades

Mode	Stage 1 Blades		Stage 2 Blades	
	Mean Frequency Hz	Standard Deviation	Mean Frequency Hz	Standard Deviation
1st Bending	267·2	2·95	288	3·13
2nd Bending	1110	21·9	1006	17·2
1st Torsion	1597	9·87	2052	12·75
3rd Bending	2753	22·5	2436	19·7
4th Bending	—	—	4560	52
2nd Torsion	4267	64·4	4872	71

having multiple restoring forces. For a rotating blade it has the form:

$$f^2 = f_0^2 + SN^2 \qquad (2.1)$$

where f is natural frequency, f_0 frequency at zero rotational speed and N the speed. If the mode shape of the blade can be determined, the Southwell coefficient S can be calculated exactly. The shape of practical blades makes exact quantitative determination of the coefficient by either experiment or analysis very difficult, but data are available to estimate values. The value of S for each mode is a function of blade geometry; the relevant parameters are aspect ratio, radius/length ratio, taper, twist and setting angle. Data in [4] for untwisted rectangular plates were used to find approximate values for S which are given in Table 2. Comparison with data in [5] and [6] indicates that the effect of neglecting taper and twist should not be serious. Radius ratio and setting angle are used explicitly in determining S; data were interpolated for applicable aspect ratios. Because the system is not quite linear, the value of S is slightly influenced by the ratio of rotational to natural frequency.

TABLE 2

Southwell Coefficients

T55 Compressor — Original Blades

	Stage 1	Stage 2
Aspect Ratio	3	4·3
Radius Ratio	1·15	0·92
Setting Angle	47·2°	48·3°
Mode	**Southwell Coefficient**	
1st Bending	2·424	2·05
2nd Bending	15·9	13·9
1st Torsion	2·63	2·25
3rd Bending	45·2	39·6
4th Bending	—	60
2nd Torsion	19·6	17·1

2.3 Frequency Scatter

Small samples (about 5–6) of each type of blade were supplied to A.R.L. for testing; measured blade frequencies exhibited significant scatter. Even if the frequency values are assumed to have a normal distribution, the small sample sizes meant that special methods were required to estimate scatter limits with their associated probability levels, the appropriate methods being those dealing with tolerance limits. Discussion of tolerance limits, and an outline of their derivation is a substantial topic and is considered in the following section.

3. LIMITS OF FREQUENCY SCATTER

3.1 Tolerance Limits

Tolerance limits need to be established when it is required to find or to control the proportion of a population having values of a particular variable within (or without) a specified range. For a variable such as the strength of a component, a single sided limit is relevant since the statistic of concern is the proportion of the population having less than a specified strength. With vibration frequency values, either upper or lower limits may define critical resonance conditions so a two-sided tolerance limit is applicable.

Two-sided tolerance limits are defined as the limits x_1, x_2 of the population variable x, within which a proportion p of the population is expected to lie. As in the present problem, statistics of population variables frequently are not known, and must be inferred from measurements on small samples. If a large series of samples were to be taken, the varying limits x_1, x_2 deduced statistically from each individual sample would encompass a proportion at least p of the population in a fraction γ of the samples. It is then concluded that the limits x_1, x_2 predicted using a single sample have a probability γ of including a proportion at least p of the population.

For known forms of probability distribution, values for tolerance limits can be derived. For a normal distribution, whose limits are symmetrical, details are given in [9]. If the distribution is not normal, results have the same form but the variable limits are changed to some

extent. Non parametric methods, not requiring any knowledge of the probability distribution, can also be used; such distribution free methods are relatively inefficient in the sense that significantly larger sample sizes are required to predict limits with comparable certainty, or to predict limit ranges at the same probability level.

3.2 Derivation of Tolerance Limits

Tolerance limits for a normally distributed population of mean value μ, variance σ^2 are derived by considering the properties of a sample of size N drawn from this population. The central limit theorem asserts that sample mean $\bar{x}$ is normally distributed with mean μ, variance σ^2/N. The sample standard deviation estimate s has a nearly normal distribution with mean σ, variance $\sigma^2/2\,(N-1)$. The statistic $\bar{x} \pm ks$ combines the distributions of $\bar{x}$ and s, and by integrating its distribution there can be found the probability γ that $\bar{x} \pm ks$ will encompass the same proportion p of the population as $\mu \pm k_p\sigma$ (k_p is the normal deviate for proportion p). This integration must be performed numerically. Since there are four variables, three of them independent, multi-dimensional interpolation (direct or inverse) may be needed to secure desired results in some cases.

A more convenient form of result is obtained by a Maclaurin series approximation as outlined in [10]. Since only the proportion of the distribution covered by a specified interval is of concern, the population may without loss of generality be standardised with mean $\mu = 0$, variance $\sigma^2 = 1$, having a standard normal distribution function:

$$f(x) = \frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}x^2}$$

Defining A by:

$$A = \int_{\bar{x}-ks}^{\bar{x}+ks} f(x)\,dx$$

Probability Pr is required such that:

$$Pr\{A \geq p\} = \gamma \qquad (3.1)$$

Define r independently of k by:

$$p = \int_{\bar{x}-r}^{\bar{x}+r} f(x)\,dx$$

Because A increases monotonically with ks, the inequality $A \geq p$ is equivalent to $ks \geq r$. Since the distributions of $\bar{x}$ and s are independent then:

$$\begin{aligned} Pr\{A \geq p \mid \bar{x}\} &= Pr\{ks \geq r \mid \bar{x}\} \\ &= Pr\{(N-1)s^2 \geq (N-1)\left(\frac{r}{k}\right)^2 \mid \bar{x}\} \end{aligned}$$

$(N-1)s^2$ has a value distributed as χ^2 for $N-1$ degrees of freedom so:

$$Pr\{A \geq p \mid \bar{x}\} = Pr\{\chi^2_{N-1} \geq (N-1)\left(\frac{r}{k}\right)^2 \mid \bar{x}\} \qquad (3.2)$$

To obtain the unconditional probability $Pr\{A \geq p\}$, $Pr\{A \geq p \mid \bar{x}\}$ is approximated by the first two terms of a Maclaurin series, noting that odd powers vanish because of symmetry:

$$Pr\{A > p \mid \bar{x}\} = Pr\{A > p \mid 0\} + \frac{\bar{x}^2}{2!} Pr''\{A > p \mid 0\} \ldots \tag{3.3}$$

As indicated $\bar{x}$ has a normal distribution function:

$$f(\bar{x}) = \sqrt{\frac{N}{2\pi}}\, e^{-\frac{1}{2}N\bar{x}^2}$$

Total probability is found by integrating over this distribution:

$$Pr\{A > p\} = \int_{-\infty}^{\infty} \left(Pr\{A > p \mid 0\} + \frac{\bar{x}^2}{2!} Pr''\{A > p \mid 0\} \ldots\right) f(\bar{x})\, d\bar{x}$$

The conditional probability terms $Pr\{A > p \mid 0\}$ and $Pr''\{A > p \mid 0\}$ within the integral are constant, so integrating $f(\bar{x})$ and its second moment:

$$Pr\{A > p\} = Pr\{A > p \mid 0\} + \frac{1}{2N} Pr''\{A > p \mid 0\} \ldots \tag{3.4}$$

Putting $\bar{x} = 1/\sqrt{N}$ in (3.3):

$$Pr\left\{A > p \mid \frac{1}{\sqrt{N}}\right\} = Pr\{A > p \mid 0\} + \frac{1}{2N} Pr''\{A > p \mid 0\} \ldots$$

Comparing with (3.4) gives:

$$Pr\{A > p\} \approx Pr\left\{A > p \mid \frac{1}{\sqrt{N}}\right\} \tag{3.5}$$

Substituting $\bar{x} = 1/\sqrt{N}$ in (3.2):

$$\gamma = Pr\{A > p\} = Pr\left\{\chi^2_{N-1} > (N-1)\left(\frac{r}{k}\right)^2 \mid \frac{1}{\sqrt{N}}\right\} \tag{3.6}$$

Corresponding values of r and k_p with $\bar{x} = 1/\sqrt{N}$ are found by equating the population fractions of the standard distribution with central and non-central limits (Fig. 1):

$$\int_{\bar{x}-r}^{\bar{x}+r} f(x)\, dx = \int_{-k_p}^{k_p} f(x)\, dx = p \tag{3.7}$$

Which as Fig. 1 shows is the same as equating the tail areas:

$$\int_{k_p}^{\bar{x}-r} f(x)\, dx = \int_{k_p}^{\bar{x}+r} f(x)\, dx$$

The integration limits are close together, so the value of $f(x)$ at the upper limits can be approximated by:

$$f(x+h) = f(x) + h f'(x) = (1 - hx) f(x)$$

since for the standard normal distribution $f'(x) = -x f(x)$. The integrals are estimated by the trapezoidal rule, and neglecting higher order terms, after manipulation there is obtained:

$$r = k_p(1 + \bar{x}^2/2) = k_p(1 + 1/2N) \tag{3.8}$$

(the latter after substituting $\bar{x} = 1/\sqrt{N}$). When this is used in (3.6) the final result becomes:

$$k = k_p \sqrt{\frac{N-1}{\chi_\gamma^2}}\,(1 + 1/2N) \tag{3.9}$$

where the value χ_γ^2 is exceeded with probability γ in a χ^2 distribution with $N-1$ degrees of freedom.

Equation (3.9) requires only readily available tables or computer algorithms for normal and χ^2 distributions. [9] verifies that despite the approximations entailed, results of (3.6) are nearly exact (even for values as small as $N = 2$, if p and γ are high). Comparison of solutions of (3.7) and (3.8) indicates that (3.9) is of similar accuracy if $N \geqq 4$.

As an example of the application of (3.9), consider a sample of size $N = 6$, probability $\gamma = 0{\cdot}95$, and population fraction $p = 0{\cdot}98$ (excludes 1% extremes). For $N - 1 = 5$ degrees of freedom χ^2 exceeds $1{\cdot}15$ at $\gamma = 0{\cdot}95$, and $k_p = 2{\cdot}326$ at $p = 0{\cdot}98$.

$$k = 2{\cdot}326\sqrt{\frac{5}{1{\cdot}15}}\left(1 + \frac{1}{12}\right) = 5{\cdot}25$$

k is tabulated as a function of N, p and γ in [8]. Figure 2 shows γ vs. p for constant $N = 5$, and Figs 3, 4 and 5 show k vs. N for various combinations of γ and p. In these plots it is seen that with large sample sizes, k asymptotically approaches Gaussian values k_p.

3.3 Properties of Frequency Data

The numerical values given in the figures apply to a normally distributed population. As noted, for other distributions numerical values of tolerance limits are changed, but generally not to a serious degree. Some of the effects of probability in the current tests can be suggested from prior knowledge.

Production control of geometric tolerances on blade dimensions would tend to reduce scatter below that of a normal distribution. Geometrical dimensions have a distribution with truncated extreme values, since blades are rejected when dimensions are outside allowable limits. Assuming that various geometrical size variations are independent of one another, and that they add together to affect the blade frequency, the central limit theorem indicates that this cumulative effect (natural frequency) will have a distribution tending more toward Gaussian than the individual contributing effects (geometrical sizes). Even though this tendency accurately represents the central features of shape of the natural frequency distribution, if the number of individual contributing effects is fairly small it will not be accurate at the extreme values which are of concern, and their distribution will remain substantially truncated relative to the Gaussian.

It is also vital to accurate estimation that the sample be random. Bias in the sample selection (e.g. if blades are all drawn from a batch which differs significantly from the population) produces correspondingly biased results, so careful sampling is important. Sampling bias results in a undirectional shift of both range limits, and truncated tolerance distributions a reduction in width of the scatter, so their combined effects tend to cancel one another at one extreme of the range, and to be unduly conservative at the other (i.e. actual scatter limits will be less than estimate).

4. EVALUATION OF RESULTS — ORIGINAL BLADES

4.1 Campbell Diagrams — Construction

Interpretation of results is facilitated by presenting data in Campbell diagrams, plotting natural frequency limits vs. rotational speed. To estimate the extent of frequency scatter, a deviation multiplier k in the tolerance limits $\bar{f} \pm ks$ was chosen to give a 95% probability of en-

compassing at least 98% of the population (i.e. all except 1% extreme values). For the sample sizes used this gave k values in the region 5–6, and frequency limits based on this criterion are given in Table 3. Figure 2 shows that for the given sample size and chosen deviation multiplier, a greater population proportion is included with lower probability or a lesser proportion with higher probability.

TABLE 3

Frequency Tolerance Limits
T55 Compressor — Original Blades

Mode	Frequency Hz Stage 1 Blades		Stage 2 Blades	
	Lower	Upper	Lower	Upper
1st Bending	252·4	282	272	304
2nd Bending	1000	1220	920	1092
1st Torsion	1548	1646	1988	2106
3rd Bending	2640	2866	2337	2535
4th Bending	—	—	4300	4820
2nd Torsion	3945	4589	4517	5227

The frequency scatter limits in Table 3, together with the Southwell coefficients in Table 2 were used in equation (2.1) to produce Campbell diagrams for first and second stage blades, shown in Figs 6 and 7 respectively. The straight lines through the origin indicate the exciting frequency produced when the blade experiences 1, 2, 3, etc. pulses during one revolution.

4.2 Discussion

The plots confirm that the "second order" vibration problem previously mentioned, refers to a possible resonance condition when the blade frequency in the 1st bending mode coincides with excitation at twice engine speed. This will occur in the range 60–78% maximum engine speed (18720 rpm) for first stage or 60–75% maximum rpm for second stage blades. Pending detailed investigation of the magnitude of the second order aerodynamic component, the RAAF was advised to avoid as far as possible operating the engine in the critical speed range [2]. Since the engine power output is relatively low at these speeds, it was fortunately possible to adopt these limitations without hampering the operational effectiveness of the helicopter.

Subsequent detailed investigation of the flow conditions at the compressor inlet showed that with the overlay screens fitted, flow distortion which would generate second order excitation was less than with the standard manufacturer approved screens [3], so the operating restriction could be removed.

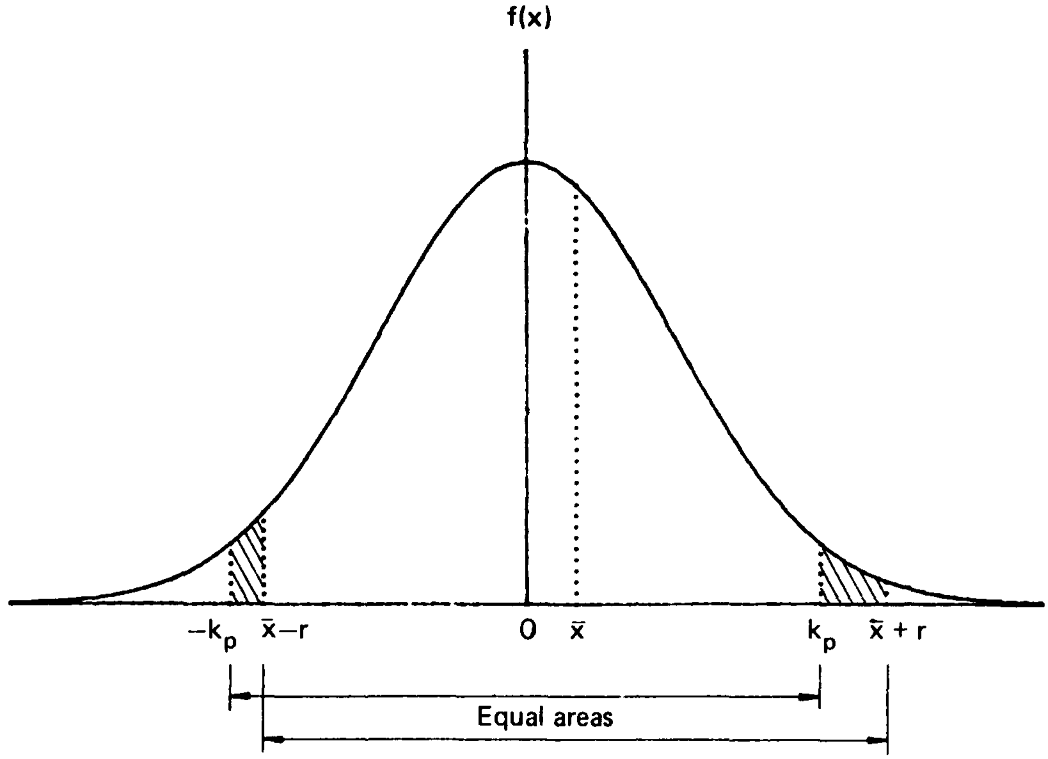


FIG. 1 CENTRAL AND NON-CENTRAL LIMITS WITH NORMAL DISTRIBUTION

Sample size N = 5

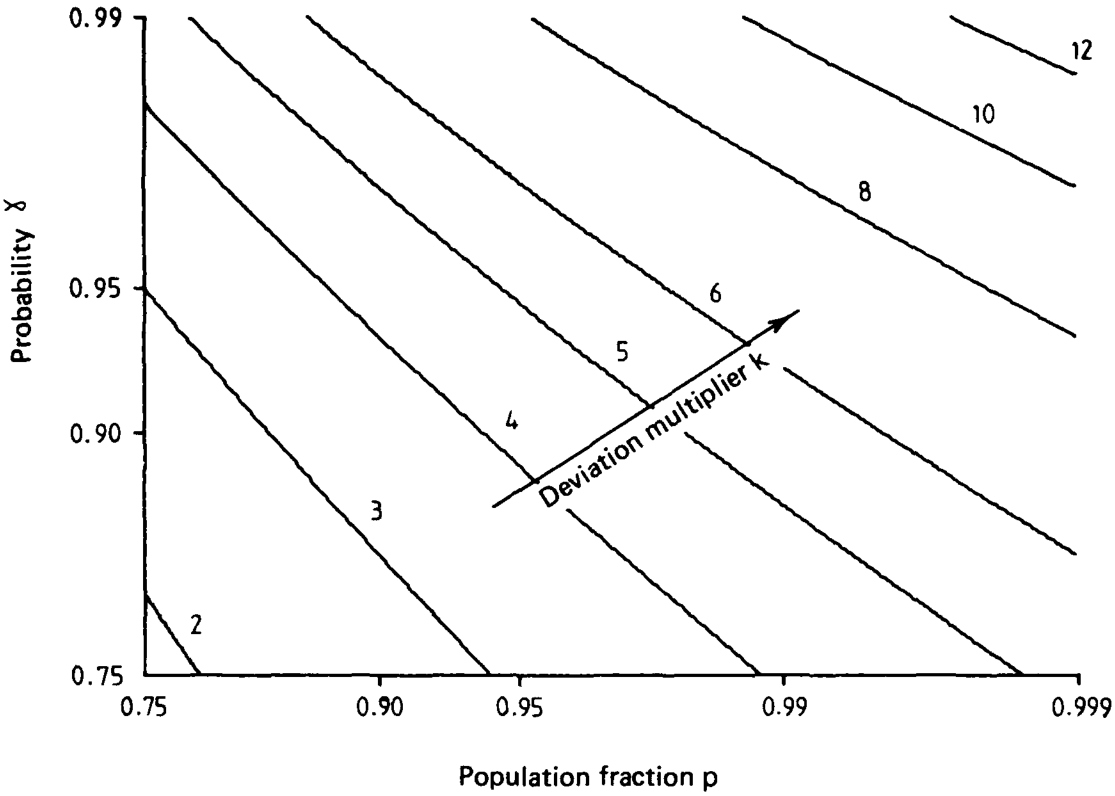


FIG. 2 TOLERANCE LIMITS

Probability $\gamma = 0.9$

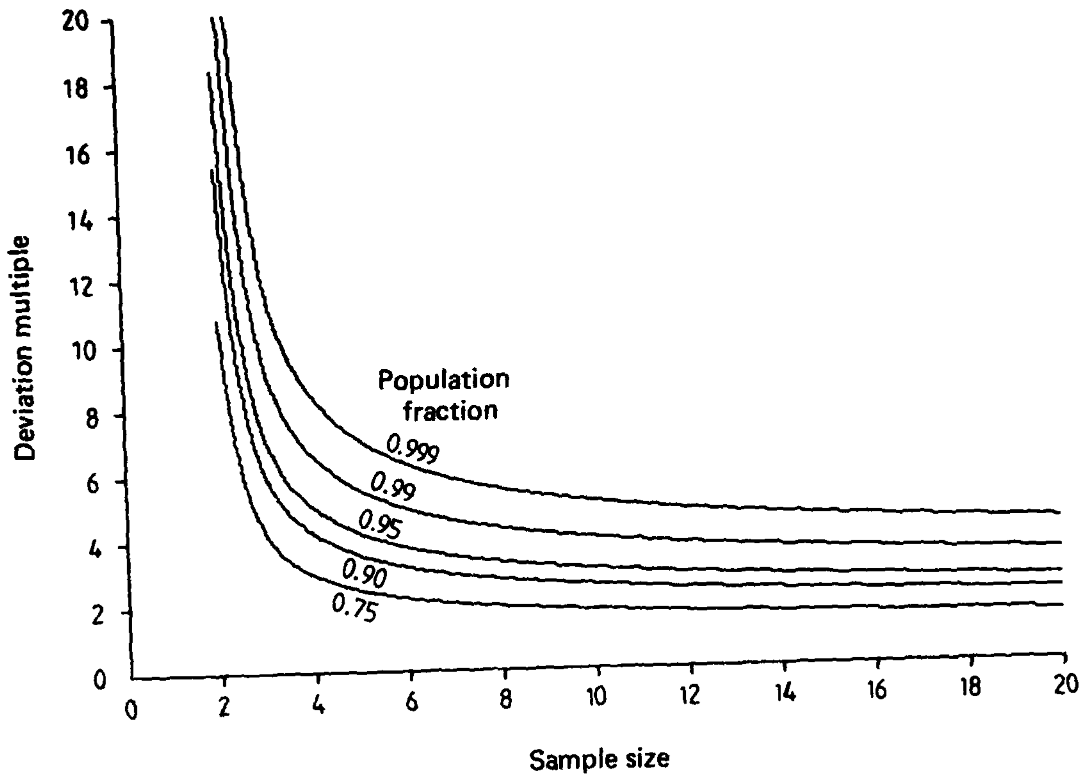


FIG. 3 TOLERANCE LIMITS

Probability γ = 0.95

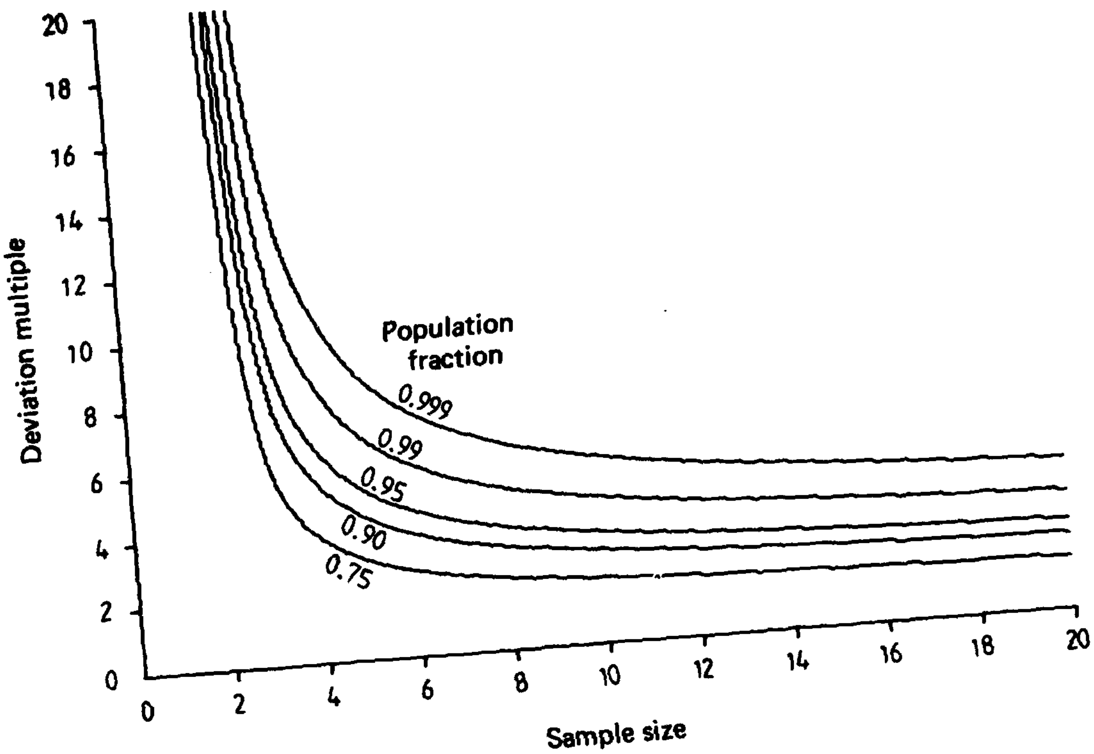


FIG. 4 TOLERANCE LIMITS

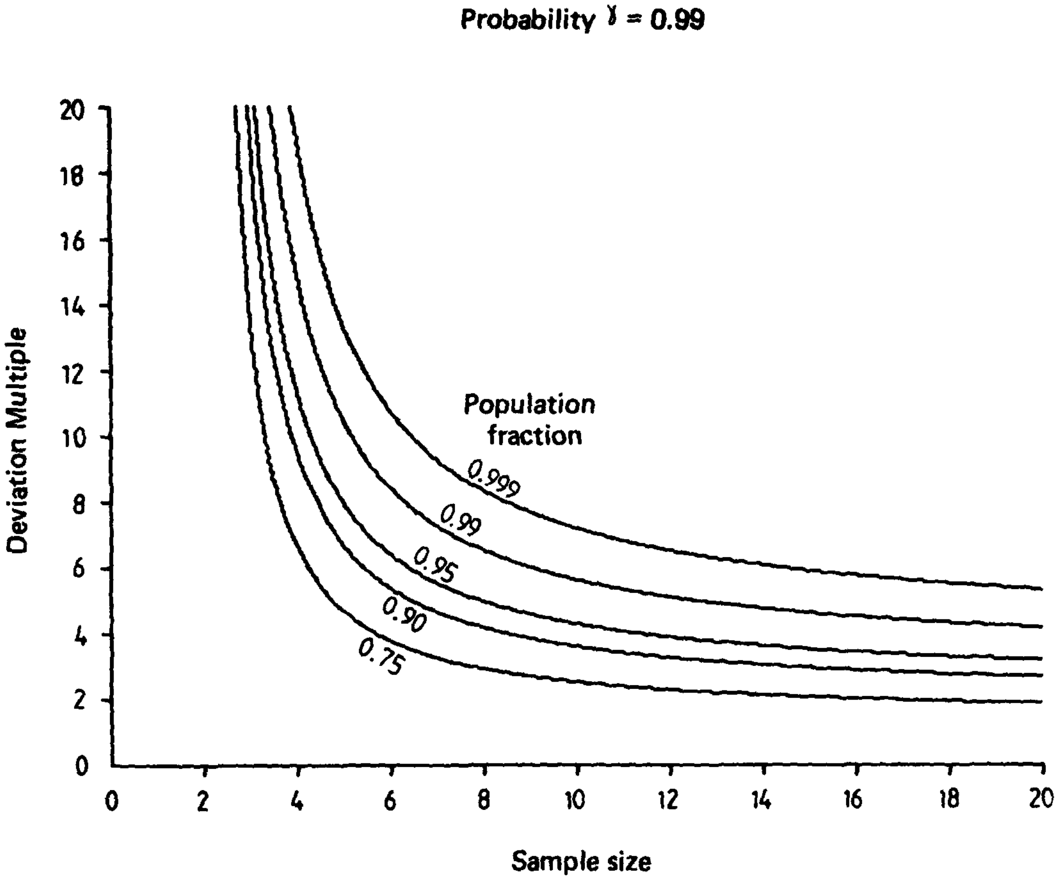


FIG. 5 TOLERANCE LIMITS

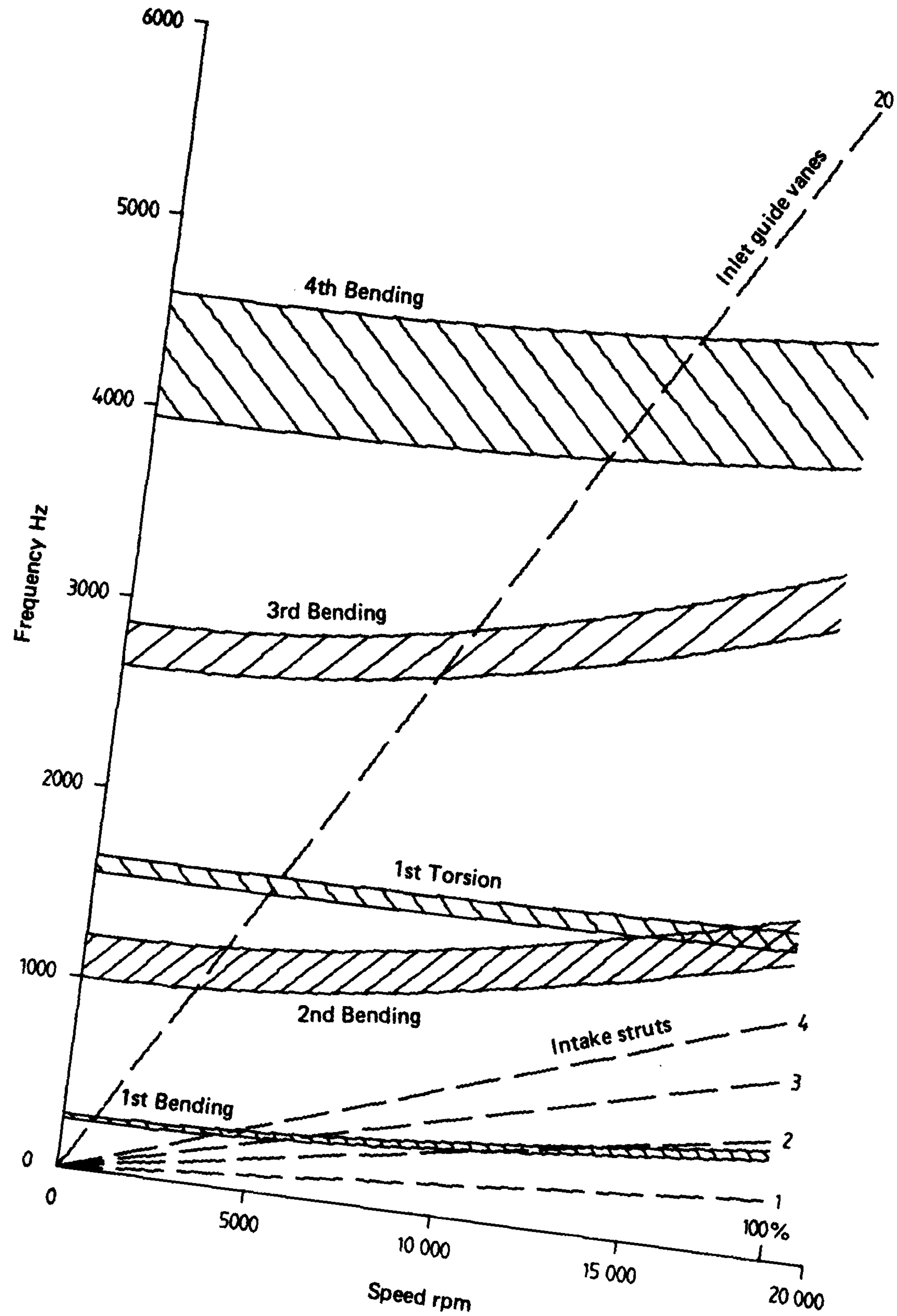


FIG. 6 T55 COMPRESSOR ORIGINAL BLADES - STAGE 1

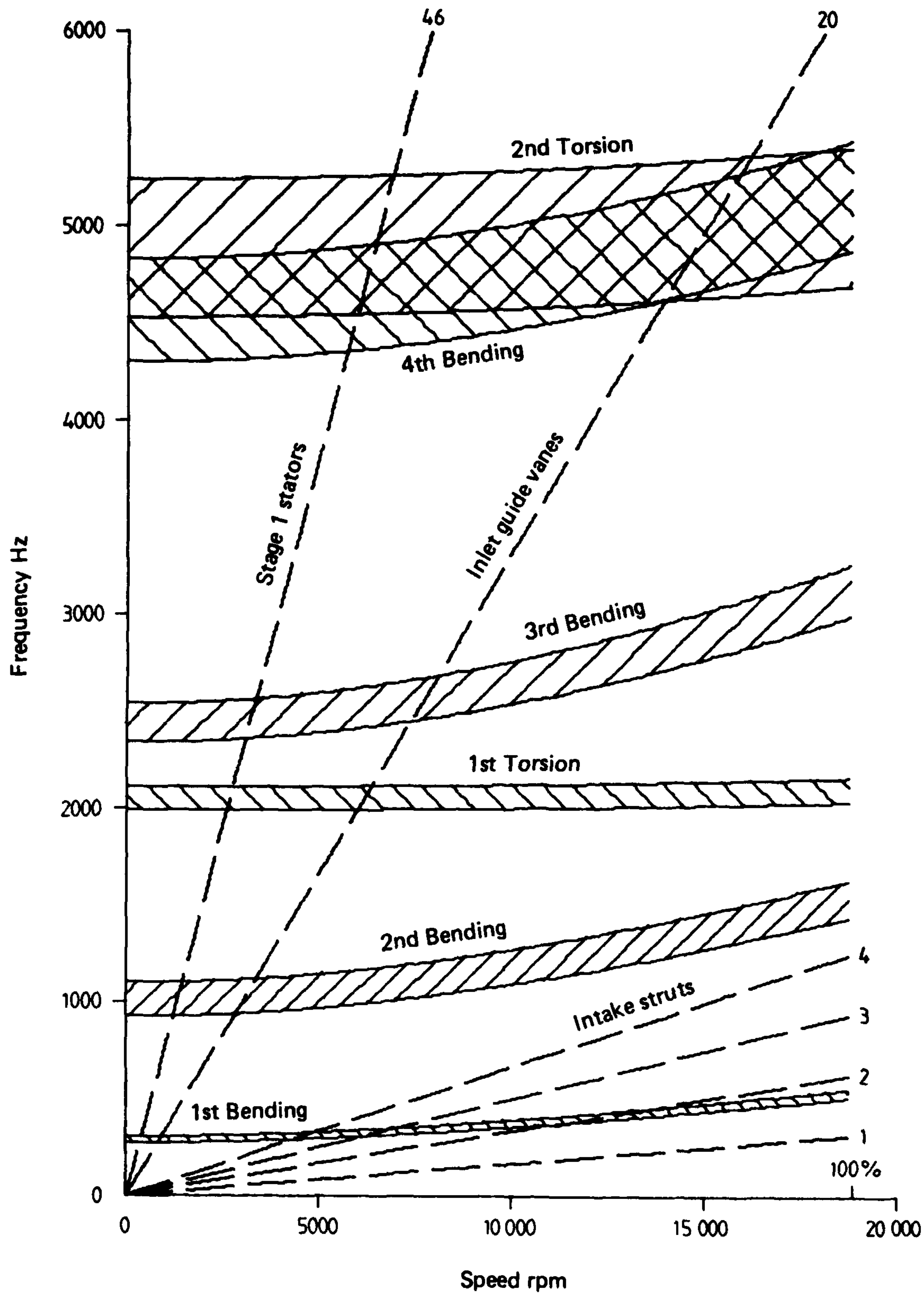


FIG. 7 T55 COMPRESSOR ORIGINAL BLADES – STAGE 2

5. WIDE CHORD BLADES

To improve engine performance and alleviate other operating problems including FOD, the engine manufacturer has produced blades of modified design for first and second stages of the compressor. The new design eliminates inlet guide vanes, and provides rotor blades which are mechanically more robust having a considerably wider chord, so they are referred to by this distinguishing feature.

The RAAF propose in due course to fit wide chord blades and this will alleviate the FOD and eliminate any associated vibration problem.

Comprehensive design validation checks were conducted by the manufacturers on the wide chord blades, and they supplied data on vibrational stresses in blades in service, as well as natural frequency data and Campbell diagrams. However to complement the earlier tests and verify the validity of test methods, frequencies were measured on a small sample of blades supplied through the RAAF. Results of frequency measurements for the wide chord blades are given in Table 4, estimated Southwell coefficients are given in Table 5, and frequency tolerance limits are given in Table 6, with Campbell diagrams plotted in Figs 8 and 9.

TABLE 4

Natural Frequencies
T55 Compressor — Wide Chord Blades

Mode	Stage 1 Blades		Stage 2 Blades	
	Mean Frequency Hz	Standard Deviation	Mean Frequency Hz	Standard Deviation
1st Bending	530	6·09	506	5·17
2nd Bending	—	—	1800	23·9
1st Edgewise	1819	7·5	—	—
1st Torsion	2265	25·2	2628	26·3
3rd Bending	—	—	5245	77·5
2nd Torsion	5220	36·4	6085	110·5
Chordwise	7380	108	—	—

TABLE 5

Southwell Coefficients
T55 Compressor — Wide Chord Blades

	Stage 1	Stage 2
Aspect Ratio	2·0	2·73
Radius Ratio	1·17	1·46
Setting Angle	54°	52°
Mode	**Southwell Coefficient**	
1st Bending	2·35	2·83
2nd Bending	—	18·55
1st Edgewise	2·66	—
1st Torsion	2·45	2·96
3rd Bending	—	52·5
2nd Torsion	17·9	22·3
Chordwise	8·12	—

TABLE 6

Frequency Tolerance Limits
T55 Compressor — Wide Chord Blades

Mode	Frequency Hz Stage 1 Blades		Frequency Hz Stage 2 Blades	
	Lower	Upper	Lower	Upper
1st Bending	495	566	475	537
2nd Bending	-	—	1657	1943
1st Edgewise	1774	1864	—	-
1st Torsion	2113	2417	2470	2786
3rd Bending	-	-	4780	5710
2nd Torsion	5002	5438	5422	6748
Chordwise	6732	8028	—	

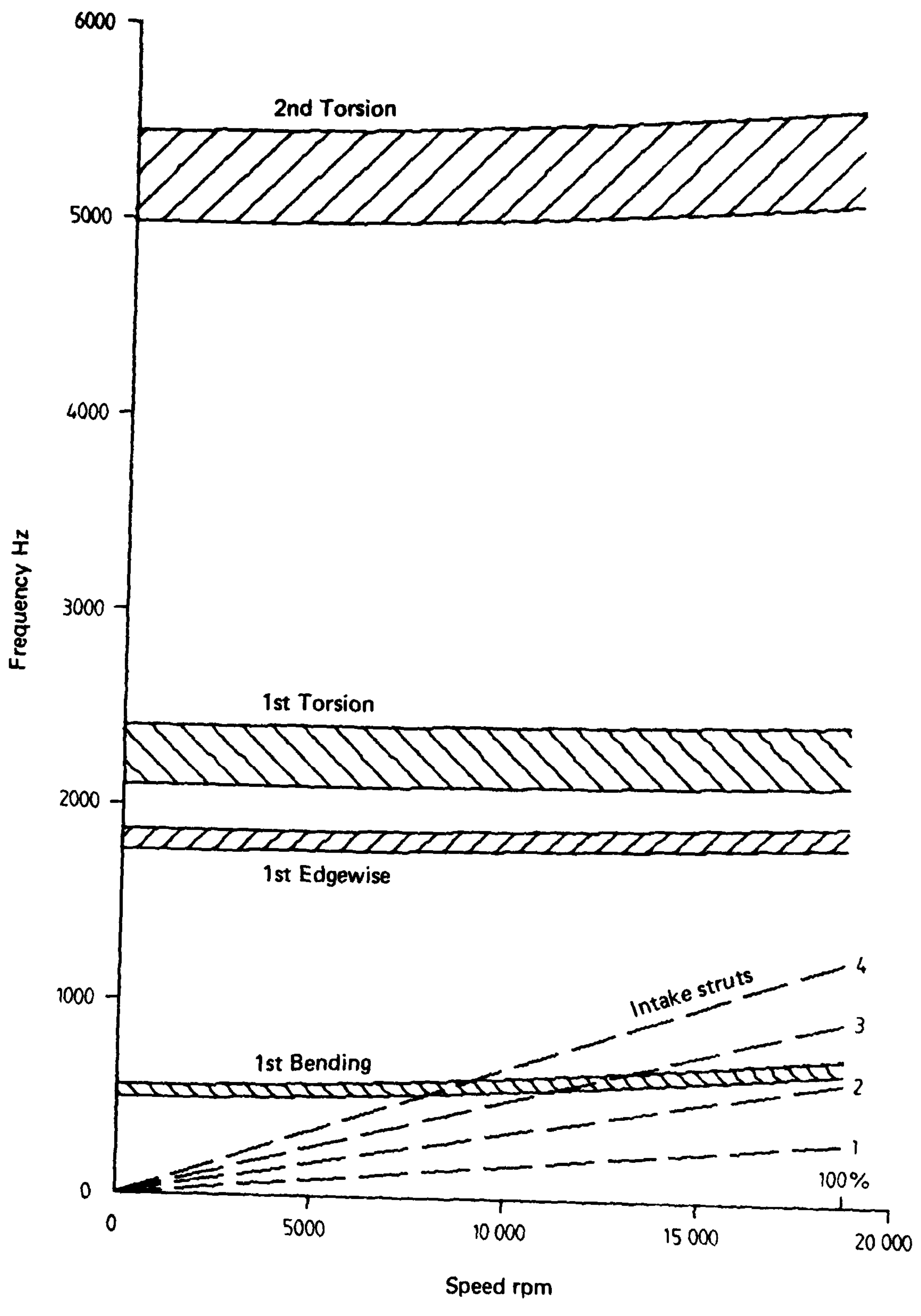


FIG. 8 T55 COMPRESSOR WIDE CHORD BLADES – STAGE 1

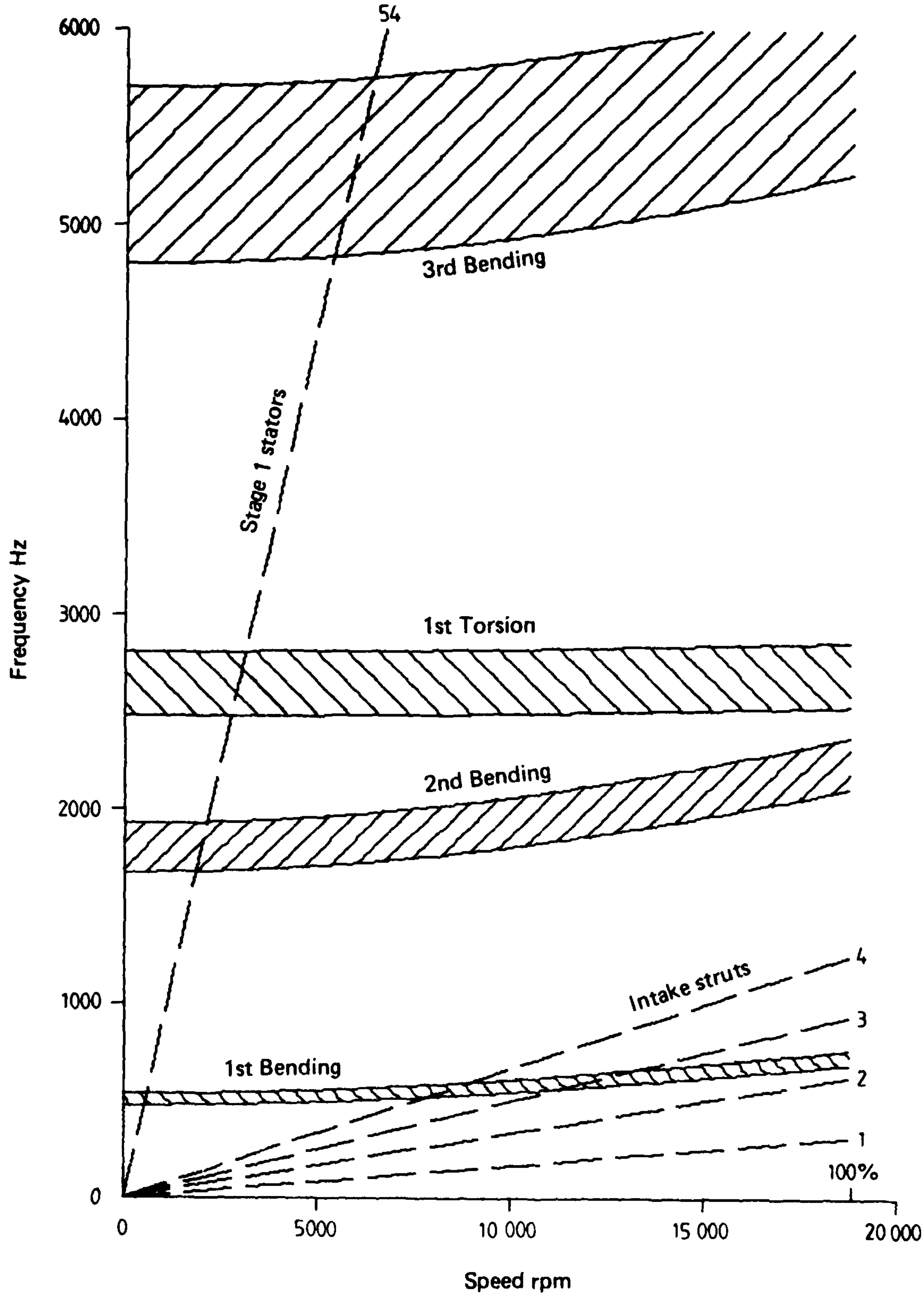


FIG. 9 T55 COMPRESSOR WIDE CHORD BLADES -- STAGE 2

Summarising these results, the wide chord blades are substantially stiffer with lowest natural frequencies nearly twice those of the original blades. Consequently any possible resonance condition with intake strut disturbances, and with stator wake disturbances, has been eliminated.

Agreement of these results and manufacturer data was very good for lower modes but some discrepancies occurred with the higher modes. These may be due to limitations of the manufacturer's vibration model, or in the experimental techniques employed in this investigation. Difficulties in the experimental area occur in producing the proper conditions of constraint, and in providing appropriate excitation for high modes (a single point excitation was used); an alternative exciter location, or multi-point excitation may have been required for some modes.

The mode designated "Chordwise" in the Stage 1 Blades (Table 3) differs from all other beam type modes encountered. Motion in this mode is mainly normal to the midsurface of the blade so the centre line of the blade section changes shape. Such plate or shell type modes are characteristic of low aspect ratio blades [7].

6 CONCLUSIONS

Tests on first and second stage compressor blades of the T55 engine confirmed that there was a potential resonance condition with second order excitation, and the extent of the resonant speed region was defined. Pending fuller investigation, operation of this region was restricted. Later investigation of inlet flow distortion showed that second order excitation, when fitted with locally designed intake screens, was less than with manufacturer approved equipment, so that no local operating restrictions were necessary.

Scatter in measured frequency values indicated that a frequency tolerance band needed to be established to adequately define the extent of any possible resonance region. As data were derived from small samples the available but infrequently used tolerance limit theory was applied, which provides an estimate of tolerance limits for a given population proportion, and corresponding probabilities. Larger sized samples provide a closer estimate, either as narrower limits, higher population proportion or higher probability, so it is desirable to consider sample size when designing the experiment.

Tests on modified wide chord blades agreed well with the manufacturers design validation data, confirming the validity of experimental techniques used. Since design specifications now require that all new equipment and components should be supported by such data, the need for tests such as reported here should only occur with equipment designed to earlier standard, and is likely to diminish in future.

REFERENCES

[1] TN006768 from AUSTAIR Washington 8 June 1976.

[2] C2/03 of 22 June 1976.

[3] M2/451 of 13 November 1978.

[4] DOKAINISH, M. A., and RAWTANI, S. (1971).—Vibration Analysis of Rotating Cantilever Plates. Intern. Journ. for Numerical Methods in Engineering, Vol. 3, 1971.

[5] HENRY, R., and LALANNE, M. (1974).—Vibration Analysis of Rotating Compressor Blades. Trans. ASME Journ. of Engineering for Industry, August 1974.

[6] BOSSAK, M. A. J., and ZIENKIEWCZ, O. C. (1973).—Free Vibration of Initially Stressed Solids with Particular Reference to Centrifugal Force Effects in Rotating Machinery. Journal of Strain Analysis, Vol. 8, No. 4, 1973.

[7] PETRICONE, R., and SISTO, F. (1971).—Vibration Characteristics of Low Aspect Ratio Compressor Blades. Trans. ASME Journ. of Engineering for Power, January 1971.

[8] HALD, A.—Statistical Theory with Engineering Applications. John Wiley, 1952.

[9] WALD, A., and WOLFOWITZ, J. (1946).—Tolerance Limits for a Normal Distribution. Annals of Mathematical Statistics, Vol. 17, 1946.

[10] KENDALL, M. G., and STUART, A.—Advanced Theory of Statistics, Vol. 2, Inference *and Relationship. Charles Griffin, London,* 1973.

DISTRIBUTION

AUSTRALIA

DEPARTMENT OF DEFENCE

Defence Central

Chief Defence Scientist
Deputy Chief Defence Scientist
Superintendent, Science and Program Administration
Controller, External Relations, Projects and Analytical Studies
} (1 copy)
Defence Science Adviser (UK) (Doc. Data sheet only)
Counsellor, Defence Science (USA) (Doc. Data sheet only)
Defence Science Representative (Bangkok)
Defence Central Library
Document Exchange Centre, DISB (18 copies)
Joint Intelligence Organisation
Librarian H Block, Victoria Barracks, Melbourne

Aeronautical Research Laboratories

Director
Library
Superintendent—Aero Propulsion
Divisional File—Aero Propulsion
G. Merrington
Author: N. Swansson

Materials Research Laboratories

Director/Library

Defence Research Centre

Library

Navy Office

Navy Scientific Adviser
RAN Aircraft Maintenance and Flight Trials Unit
Directorate of Naval Aircraft Engineering
Superintendent, Aircraft Maintenance and Repair

Army Office

Scientific Adviser—Army

Air Force Office

Air Force Scientific Adviser
Aircraft Research and Development Unit
  Scientific Flight Group
  Library
Technical Division Library

Director General Aircraft Engineering—Air Force
Director General Operational Requirements—Air Force
HQ Operational Command (SMAINTSO)
HQ Support Command (SLENGO)
RAAF Academy, Point Cook

Government Aircraft Factories
Manager
Library

DEPARTMENT OF AVIATION
Library

STATUTORY AND STATE AUTHORITIES AND INDUSTRY
Trans-Australia Airlines, Library
Qantas Airways Limited
SEC of Vic., Herman Research Laboratory, Library
Ansett Airlines of Australia, Library
Commonwealth Aircraft Corporation, Library
Hawker de Havilland Aust. Pty. Ltd., Bankstown, Library
Rolls Royce of Australia Pty. Ltd., Mr C. G. A. Bailey

Universities and Colleges

Adelaide	Barr Smith Library
	Professor of Mechanical Engineering
Flinders	Library
La Trobe	Library
Melbourne	Engineering Library
Monash	Hargrave Library
Newcastle	Library
New England	Library
Sydney	Engineering Library
NSW	Physical Sciences Library
	Professor R. A. A. Bryant, Mechanical Engineering
Queensland	Library
Tasmania	Engineering Library
Western Australia	Library
	Associate Professor J. A. Cole, Mechanical Engineering
RMIT	Library

CANADA
International Civil Aviation Organization, Library
NRC
Aeronautical & Mechanical Engineering Library
Division of Mechanical Engineering, Director

Universities and Colleges

Toronto	Institute for Aerospace Studies

CZECHOSLOVAKIA
Aeronautical Research and Test Institute (Prague), Head

FRANCE

ONERA, Library

INDIA

Defence Ministry, Aero Development Establishment, Library
Gas Turbine Research Establishment, Director
Hindustan Aeronautics Ltd., Library
National Aeronautical Laboratory, Information Centre

ISRAEL

Technion-Israel Institute of Technology
Professor J. Singer
Dr I. T. Bar-Itzhack

JAPAN

National Aerospace Laboratory
Institute of Space and Astronautical Science, Library

NETHERLANDS

National Aerospace Laboratory (NLR), Library

NEW ZEALAND

Defence Scientific Establishment, Library
RNZAF, Vice Consul (Defence Liaison)

Universities

Canterbury	Library
	Professor D. Stevenson, Mechanical Engineering

SWEDEN

Aeronautical Research Institute, Library
Swedish National Defence Research Institute (FOA)

UNITED KINGDOM

Ministry of Defence, Research, Materials and Collaboration
CAARC, Secretary
Royal Aircraft Establishment
Bedford, Library
Pyestock North, Director
Admiralty Research Establishment
St Leonard's Hill, Superintendent
National Engineering Laboratory, Library
British Library, Lending Division
GEC Gas Turbines Ltd., Managing Director
Ricardo & Company Engineers (1927) Ltd., Manager
Rolls-Royce Ltd., Aero Division Bristol, Library
British Aerospace
Kingston-upon-Thames, Library
Hatfield-Chester Division, Library

British Hovercraft Corporation Ltd., Library
Short Brothers Ltd., Technical Library

Universities and Colleges

Bristol	
Bristol	Engineering Library
Cambridge	Library, Engineering Department
	Whittle Library
London	Professor G. J. Hancock, Aero Engineering
Nottingham	Science Library
Southampton	Library
Strathclyde	Library
Cranfield Inst. of Technology	Library
Imperial College	Aeronautics Library

UNITED STATES OF AMERICA

NASA Scientific and Technical Information Facility
Boeing Vertol
United Technologies Corporation, Library
Lockheed-California Company
Lockheed Missiles and Space Company
Lockheed Georgia
McDonnell Aircraft Company, Library
Avco Lycoming

Universities and Colleges

Florida	Aero Engineering Department
	Professor D. C. Drucker
Johns Hopkins	Professor S. Corrsin, Engineering
Iowa State	Dr G. K. Serovy, Mechanical Engineering
Iowa	Professor R. I. Stephens
Princeton	Professor G. L. Mellor, Mechanics
Massachusetts Inst. of Technology	MIT Libraries

SPARES (10 copies)

TOTAL (139 copies)

Department of Defence

DOCUMENT CONTROL DATA

1. a. AR No.	AR-004-046
1. b. Establishment No.	ARL-AERO-PROP-R-168
2. Document Date	September 1985
3. Task No.	DST 82/041
4. Title	INVESTIGATION OF BLADE VIBRATION T55-L-11C COMPRESSOR STAGES 1 AND 2
5. Security a. document	Unclassified
b. title	U
c. abstract	U
6. No. Pages	22
7. No. Refs	9
8. Author(s)	N. S. Swansson
9. Downgrading Instructions	--
10. Corporate Author and Address	Aeronautical Research Laboratories P.O. Box 4331, Melbourne Vic., 3001.
11. Authority (*as appropriate*)	a. Sponsor b. Security c. Downgrading d. Approval
12. Secondary Distribution (*of this document*)	Approved for public release

Overseas enquirers outside stated limitations should be referred through ASDIS, Defence Information Services Branch, Department of Defence, Campbell Park, CANBERRA, ACT, 2601.

13. a. This document may be ANNOUNCED in catalogues and awareness services available to . . .

No limitations

13. b. Citation for other purposes (*i.e. casual announcement*) may be (*select*) unrestricted (*or*) as for 13 a.

14. Descriptors		15. COSATI Group
Vibration	Tolerances (Statistics)	21050
Resonant frequency		20110
Forced vibration		
Compressor blades		

16. Abstract

A vibration survey of the first and second stage compressor blades on the T55-L-11C engine was conducted following the manufacturer's warning that the proposed fitting of locally designed intake screens might lead to blade vibration problems. Vibration tests and a simple analysis indicated a possible resonance region. Special statistical methods for small samples were used to fully define tolerance limits for this region and establish confidence levels in the limits.

Operators were initially advised to restrict operation in the potential resonance region. Subsequent intake flow tests demonstrated that the screens generated no abnormal excitation, so the restrictions could be discontinued.

This page is to be used to record information which is required by the Establishment for its own use but which will not be added to the DISTIS data base unless specifically requested.

16. Abstract (Contd)		
17. Imprint Aeronautical Research Laboratories, Melbourne		
18. Document Series and Number Aero Propulsion Report 168	**19. Cost Code** 47 7455	**20. Type of Report and Period Covered**
21. Computer Programs Used		
22. Establishment File Ref(s)		

END
DTIC
9 — 86